#6

PATENT

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

IN RE APPLICATION OF	:	John Henry May
FOR	:	**FLOW CONTROL VALVE**
SERIAL NO.	:	10/022,784
FILED	:	December 17, 2001
EXAMINER	:	Unknown
ART UNIT	:	3753
ATTORNEY DOCKET NO.	:	UDL 2 0018

TRANSMITTAL LETTER OF 35 U.S.C. § 119 FOREIGN PRIORITY CLAIM FOREIGN PRIORITY DOCUMENT

Assistant Commissioner for Patents
Washington, D.C. 20231

Dear Sir:

Applicant hereby claims priority under 35 U.S.C. § 119 for the above-identified U.S. patent application. This claim of priority is based upon **British Application No. GB-0030786.8, filed December 18, 2000.** A certified copy of this prior British application is enclosed.

Respectfully submitted,

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CERTIFICATE OF MAIL

I hereby certify that this Transmittal Letter of Foreign Priority Document and accompanying papers are being deposited with the United States Postal Service as FIRST CLASS MAIL addressed to the Assistant Commissioner For Patents, Washington, D.C. 20231 on March 18, 2002.

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Cardiff Road
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1.	Your reference	SMR/TLB/P550066
2.	Patent application number *(The Patent Office will fill in this part)*	**0030786.8** 18 DEC 2000
3.	Full name, address and postcode of the or of each applicant *(underline all surnames)*	Fluid Controls UK Ltd Unit 2 29 Clophill Road Maulden Beds MK45 2AA
	Patents ADP number *(if you know it)*	7629231001
	If the applicant is a corporate body, give the country/state of its incorporation	UK
4.	Title of the invention	FLOW CONTROL VALVE
5.	Name of your agent *(if you have one)*	Urquhart-Dykes & Lord
	"Address for service" in the United Kingdom to which all correspondence should be sent *(including the postcode)*	Midsummer House 411C Midsummer Boulevard Central Milton Keynes, MK9 3BN United Kingdom
	Patents ADP number *(if you know it)*	1644008

6. If you are declaring priority from one or more earlier patent applications, give the country and the date of filing of the or of each of these earlier applications and *(if you know it)* the or each application number

Country	Priority application number *(if you know it)*	Date of filing *(day / month / year)*

7. If this application is divided or otherwise derived from an earlier UK application, give the number and the filing date of the earlier application

Number of earlier application	Date of filing *(day / month / year)*

8. Is a statement of inventorship and of right to grant of a patent required in support of this request? *(Answer 'Yes' if:*
a) any applicant named in part 3 is not an inventor, or
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Description	9
Claim(s)	2
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Drawing(s)	7

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Priority documents	-
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Statement of inventorship and right to grant of a patent *(Patents Form 7/77)*	-
Request for preliminary examination and search *(Patents Form 9/77)*	-
Request for substantive examination *(Patents Form 10/77)*	-
Any other documents *(please specify)*	

11. I/We request the grant of a patent on the basis of this application.

Signature Urquhart-Dykes Date 18.12.00
URQUHART-DYKES & LORD

12. Name and daytime telephone number of person to contact in the United Kingdom — Mr Simon Raynor - 01908 666645

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DUPLICATE

FLOW CONTROL VALVE

The present invention relates to a flow control valve for delivering a substantially constant flow rate of fluid irrespective of the differential pressure across the valve.

For convenience, the invention will be described with particular reference to an application in a re-circulating hot water radiator central heating system, but it will be appreciated that its uses are not so limited and, indeed, it has wide applicability in fluid systems generally.

It is conventional practice to use calibration valves to balance the distribution of flows in large central heating systems such as in a multi-storey office block. A primary pipe loop re-circulates pumped heated water from the boiler usually located in the basement to the uppermost floor. At each floor a secondary piping loop is directly connected to the primary loop and feeds a series of radiators connected between the supply and return pipes of the secondary loop. Tertiary pipe loops may also be connected to secondary loops, and so forth. Clearly, the differential pressure across any pipe loop is dependent upon the height of the entry and exit points from the boiler and the individual pipe run friction losses. In addition to the difficulties caused by this, the flows required to meet the heating requirements on each floor are not necessarily the same and indeed may even change on a daily basis as a function of individual requirements (turning radiators on and off) or occupancy.

To obtain the desired distribution of flows necessitates the use of balancing valves, which are usually fitted between the last radiator and the connection from the return pipe of the secondary loop to the primary loop, or the tertiary pipe loop to the secondary and so forth. To set the design flow manually the degree of throttling of any particular balancing valve necessitates that a flow meter is also installed in the pipe loop. The flow meter is usually but not exclusively connected by way of differential pressure tappings across the balancing valve. However, adjusting the setting of any one valve affects the differential pressure across all the other valves, thus manual adjustment is both time-consuming and inaccurate. Furthermore, if a change occurs in either the head flow characteristic of the pump or the

individual friction resistance of any of the individual pipes, this too may alter the optimal setting of one or all of the balance valves.

An alternative and preferable approach is to use constant flow valves. A common arrangement uses a variable orifice set against a spring so that the differential pressure determines the degree of occlusion across the variable orifice. In one such variable orifice type valve, sold by Hattersley Newman Hender Limited under the trademark AUTOFLOW, the variable orifice is formed in the side wall of a spring-biased piston, which moves relative to a sleeve according to the differential pressure. The orifice area is divided into a front facing fixed orifice and one or more side orifices such that the combined variable discharge area yields the design flow over the required range of differential pressures. This yields both primary and secondary flow paths. When the differential pressure is low, a large discharge area is provided and when the differential pressure is high, the spring is compressed and the sleeve partially occludes the orifice, thereby maintaining a substantially constant flow rate. The piston and the spring may be provided in the form of a cartridge that can be removed from the main valve body and replaced with another cartridge providing a different flow rate and/or different pressure range.

A number of problems exist with this arrangement: first, for low and very low flows the Reynolds numbers are in the lamina or transitional regime of flows, which can cause a lack of repeatability due to the variability in the profile of the approach flow. Second, the variable occlusions machined in the side walls to provide the required constant flow rates necessitate very accurate machining. In conventional form this approach also necessitates that an individual and precise geometry of the variable occlusions is required for any given flow. Owing to the existence of one or more flow paths through the piston orifices, the division of flow between the paths is not necessarily repeatable and therefore this arrangement tends to lead to hysteresis between rising and falling secondary pipe resistances. This can result in the flowrate tolerance being outside the industry expected limits of ± 5%.

Another flow control valve described in US 3,464,439 (Budzich) has a resiliently-biased piston mounted for sliding movement in a cylinder. The piston has an inlet opening in its end face and a number of outlet openings in its side wall. The inlet opening is partially

occluded by a tapered probe that extends through the opening, leaving an annular flow passageway. The outlet openings are also partially occluded by the walls of the cylinder. The position of the piston depends on the differential pressure across the valve, the degree of occlusion of both the inlet and outlet openings increasing as the differential pressure increases.

The flow path through the valve is complicated leading to unpredictable flow patterns and poor flow control, particularly at low differential pressures. The device also relies on the use of two sets of shaped apertures, requiring complicated and difficult machining operations, and is mechanically complex.

WO 00/03597 (May) describes an adjustable flow control valve including a resiliently-biased piston and an adjustable throttle plate that is positioned adjacent one edge of the piston. The distance between the throttle plate and the piston can be adjusted to adjust the flow rate through the valve. The valve is mechanically complex and requires the use of complicated manufacturing processes.

It is an object of the present invention to provide a flow control valve that mitigates at least some of the disadvantages associated with the previous flow control valves, as described above.

According to the present invention there is provided a flow control valve including a body member having a bore defining a fluid flow passageway, a resiliently-biased piston member mounted in said passageway for movement relative to the body member in response to the differential fluid pressure across the valve, said piston member defining an annular throttling orifice between said piston member and said bore, wherein at least a portion of said passageway has a non-uniform cross-section, such that the size of the annular orifice depends on the position of the piston member relative to the body member.

The valve provides a substantially constant fluid flow rate across a wide range of differential pressures, including very low differential pressures when the flow is in the lamina or transitional regime. The valve is also mechanically simple, and is easy to manufacture and reliable in operation.

The flow rate of fluid through the valve is of course substantially constant only for variations in the differential pressure that lie within a predetermined range: i.e. between upper and lower operational limits, for example from 10kPa to 250kPa, or from 30kPa to 450kPa, depending on the chosen design characteristics of the valve. The statement that the flow rate is "substantially constant" implies that the flow rate is regulated to within a tolerance of, for example, ±5%.

The valve does not rely upon the use of one or more precisely machined geometrically complex shaped side orifices and can therefore be manufactured more cheaply than existing constant flow valves. Also, as the valve only requires a single orifice, the hysteresis effects caused by cascading flows are avoided.

Advantageously, the non-uniform portion of the fluid flow passageway increases in size towards an inlet end of said passageway, and is preferably flared or trumpet-shaped.

Advantageously, the piston member includes a piston head, and said throttling aperture is defined between a downstream edge of said piston head and said non-uniform portion of the fluid flow passageway. The piston head may be substantially cylindrical.

Advantageously, the piston head has a side wall that defines with the non-uniform portion of the fluid flow passageway an annular fluid flow slot, wherein the length and the area of said annular slot depend on the position of the piston member relative to the body member.

The use of frictional flow resistance as provided by the annular slot gives improved flow control at low flow rates.

Advantageously, the piston member includes a support structure, said support structure being mounted for sliding movement in the bore. The piston head may be connected to the support structure for movement therewith and extends from said support structure towards an inlet end of said valve. The support structure may include a substantially axial fluid flow passageway. The support structure may be engaged by a resilient biasing member.

The flow control valve may include a housing in which the body member can be mounted, wherein said housing is capable of accommodating interchangeable flow control valve cartridges having different fluid flow capacities.

An embodiment of the invention will now be described, by way of example, with reference to the accompanying drawings, in which:

Figure 1 is a partially sectional side view of an assembled flow control valve;

Figure 2 is a sectional side view of a main body member;

Figure 3 is a side view of a piston member;

Figure 4 is a section on line IV-IV of Figure 3;

Figure 5 is a top view of the piston member;

Figure 6 is an isometric view of a bottom ring member;

Figures 7, 8 and 9 are sectional side views showing the valve in a fully open condition, an intermediate open position and fully closed;

Figure 10 is a schematic side section illustrating the flow of fluid through the valve, and

Figure 11 is a partial sectional side view, illustrating typical trumpet sizes vs. flow rate.

The constant flow valve is constructed in the form of a cartridge that, in use, is mounted in a housing (not shown). The valve includes a main body member 1, a piston member 2, a compression spring 3 and a bottom ring 4. The dimensions of these components may vary, to provide different predetermined fluid flow rates, and the housing may be capable of accommodating a range of different cartridges, according to the required flow rate.

The main body 1 is substantially cylindrical having an inlet end 6 and a outlet end 8. An axial bore 10 that defines a fluid flow passageway extends longitudinally through the main body, the bore including an upper portion 10a, a middle portion 10b and a lower portion 10c.

The upper portion 10a of the bore is of non-uniform diameter and increases in diameter towards the inlet end 6. This upper portion is flared or trumpet-shaped and, in the example shown in the drawings, it increases in diameter from approximately 13mm at its lower end to approximately 15mm at the inlet end. The shape of the flared bore is defined by a polynomial progression, as in the bell of a trumpet.

The middle portion 10b of the bore has a uniform diameter, which in the example is approximately 15.5mm. This provides an annular step 12 at the junction between the middle portion 10b and the upper portion 10a.

The lower portion 10c is substantially the same diameter as the middle portion 10b, but it includes a screw thread 14 that is cut into the cylindrical wall of the bore.

The external surface 16 of the body member is substantially cylindrical, in the example having a diameter of approximately 20.5mm. A reduced diameter portion 17 having a diameter of approximately 20mm is provided towards the inlet end 6, and a groove 18 having a diameter of approximately 19mm extends circumferentially around the middle of the external cylindrical surface 16. The reduced diameter portion 17 and the groove 18 are used for mounting the body member 1 in a housing, the groove 18 accommodating an O-ring (not shown) for sealing the valve against leaks.

The piston member 2 includes a solid cylindrical head 22 having a circular end face 23 and a cylindrical side wall 24, which is connected by two legs 25 to a support structure 26. In the example, the head 22 has an outside diameter of approximately 12.5mm and a length of about 6.5mm.

The outside diameter of the head 22 is slightly less than the minimum diameter of the flared upper bore portion 10a and is located within, or just above, the inlet end 6 of the fluid flow passageway, to define an annular throttling orifice 28 between the flared wall of the upper bore portion 10a and the lower edge 30 of the head 22. The area of this orifice depends on the position of the piston member 2 relative to the body member 1.

In addition, the cylindrical side wall 24 of the piston head 22 and the flared wall of the upper bore portion 10a define an annular fluid flow slot 31, the length and cross-sectional area of which depend on the position of the piston member 2 relative to the body member 1. In the example, the length of this slot can vary from 0mm to 6.5mm.

The support structure 26 includes an upper portion 32, a middle portion 34 and lower portion 36. An axial bore 38 extends through the support structure 26, to provide a fluid flow passageway.

The support structure 26 is located within the middle bore portion 10b of the body member 1. The upper portion 32 has an outside diameter that is fractionally less than the internal diameter of the middle bore portion 10b, allowing the piston member 2 to slide longitudinally relative to the main body member 1. Upwards movement of the piston member 2 is limited by the upper portion 32 of the support structure 26 engaging the step 12 in the body member 1, whereas downwards movement is limited by engagement with the bottom ring 4.

The middle portion 34 of the support structure 26 has an outside diameter slightly greater than the internal diameter of the compression spring 3, which has a push fit over that portion. The lower portion 36 has a slightly smaller diameter, to extend loosely through the coils of the spring 3.

The bottom ring 4 is annular and has an external screw thread 40 that engages the internal screw thread 14 in the lower bore portion 10c. A flange 42 extends inwards at the lower end of the bottom ring 4, to provide a seat for the lower end of the spring 3. Two diametrically opposed notches 44 are provided in the flange 42, for engagement by a tightening tool.

In the assembled flow control valve, at very low differential pressures the piston member 2 is biased upwards by the compressed spring 3 to the fully open position shown in figure 7, in which the lower edge 30 of the piston head is approximately level with the inlet end 6 of the upper bore portion 10a. In use, fluid flows through the valve from the inlet end 6 to the outlet end 8. The fluid flows past the piston head 22 through the annular throttling orifice 28 between the lower edge 30 of the piston head 22 and the flared upper bore portion 10a. The fluid then passes through the bore 38 in the piston support structure 26 and the middle and lower bore portions 10b,10c in the body member 1 before exiting the valve through the outlet end 8.

When the differential pressure across the valve increases, the piston member 2 is depressed to an intermediate open position, compressing the spring 3, as shown in figure 8. The length of the annular slot 31 between the cylindrical wall of the piston head 22 and the upper bore portion 10a is thus increased, and the cross-sectional area of that slot is decreased. At the same time, the cross-sectional area of the annular throttling orifice 28

between the lower edge 30 of the piston head 22 and the flared upper bore portion 10a is reduced.

With further increases in the differential pressure, the piston member 2 is depressed further, until it reaches a fully closed position, as shown in figure 9. The length of the annular slot 31 between the cylindrical wall of the piston head 22 and the upper bore portion 10a is further increased, and the cross-sectional area of that slot is further decreased and the cross-sectional area of the annular throttling orifice 28 is further reduced.

As the fluid flows through the annular slot 31, frictional losses are incurred, which tend to restrict the flow of fluid through the valve. These frictional losses are proportional to the length of the slot and inversely proportional to its length, and therefore increase with the differential pressure across the valve, as the piston member 2 is depressed.

Further, as the fluid flows through the throttling orifice 28, there is a sudden drop in fluid flow speed, leading to a pressure drop across the orifice. This pressure drop is inversely proportional to the cross-sectional area of the orifice, and therefore increases with the differential pressure across the valve, as the piston member 2 is depressed.

The flow of fluid through the valve is illustrated in figure 10, which also includes an expression relating to the flowrate of fluid through the valve.

The valve therefore reacts to changes in the differential pressure by opening or closing, to maintain a substantially constant flow rate of fluid through the valve, the flow being controlled both by the pressure drop across the annular orifice 28, and the frictional losses in the annular passageway 31 between the cylindrical wall of the piston head 22 and the wall of the upper bore portion 10a. The combination of these two effects has been found to provide a very stable flow rate across a wide range of differential pressures, including very low differential pressures.

Various modifications of the invention are possible, some examples of which will now be described.

The valve may be manufactured in different sizes, to provide different flow rates. Figure 11 includes a table showing typical trumpet sizes for different designed flow rates.

The strength of the spring may also be varied to provide different designed flow rates. The spring may be replaced by another resilient biasing member, for example an elastomeric material or a cylinder of compressed gas. Although the valve is preferably made of stainless steel, it may also be made of other materials including plastics, ceramics and composites. Different methods of manufacture may also be employed, including for example investment casting and die casting.

Claims

1. A flow control valve including a body member having a bore defining a fluid flow passageway, a resiliently-biased piston member mounted in said passageway for movement relative to the body member in response to the differential fluid pressure across the valve, said piston member defining an annular throttling orifice between said piston member and said bore, wherein at least a portion of said passageway has a non-uniform cross-section, such that the size of the annular orifice depends on the position of the piston member relative to the body member.

2. A flow control valve according to claim 1, wherein the non-uniform portion of the fluid flow passageway increases in size towards an inlet end of said passageway.

3. A flow control valve according to claim 2, wherein the non-uniform portion of the fluid flow passageway is flared.

4. A flow control valve according to any one of the preceding claims, wherein said piston member includes a piston head, and said throttling aperture is defined between a downstream edge of said piston head and said non-uniform portion of the fluid flow passageway.

5. A flow control valve according to claim 4, wherein the piston head is substantially cylindrical.

6. A flow control valve according to claim 4 or claim 5, wherein the piston head has a side wall that defines with the non-uniform portion of the fluid flow passageway an annular fluid flow slot, wherein the length and the area of said annular slot depend on the position of the piston member relative to the body member.

7. A flow control valve according to any one of claims 4 to 6, wherein the piston member includes a support structure, said support structure being mounted for sliding movement in the bore.

8. A flow control valve according to claim 7, wherein the piston head is connected to the support structure for movement therewith and extends from said support structure towards an inlet end of said valve.

9. A flow control valve according to claim 7 or claim 8, wherein the support structure includes a substantially axial fluid flow passageway.

10. A flow control valve according to any one of claims 7 to 9, wherein the support structure is engaged by a resilient biasing member.

11. A flow control valve according to any one of the preceding claims, including a housing in which the body member can be mounted, wherein said housing is capable of accommodating interchangeable flow control valve cartridges having different fluid flow capacities.

12. A flow control valve substantially as described herein with reference to and as illustrated by the accompanying drawings.

Abstract

FLOW CONTROL VALVE

A flow control valve includes a body member (1) having a bore (10) defining a fluid flow passageway. A resiliently-biased piston member (2) is mounted in said passageway for movement relative to the body member (1) in response to the differential fluid pressure across the valve. The piston member (2) defines an annular throttling orifice (28) between said piston member and said bore. At least a portion (10a) of the passageway has a non-uniform cross-section, such that the size of the annular orifice (28) depends on the position of the piston member relative to the body member. (Figure 1)

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ITEM No	DESCRIPTION	QTY
1	MAIN BODY	1
2	PISTON	1
3	SPRING	1
4	BOTTOM RING	1

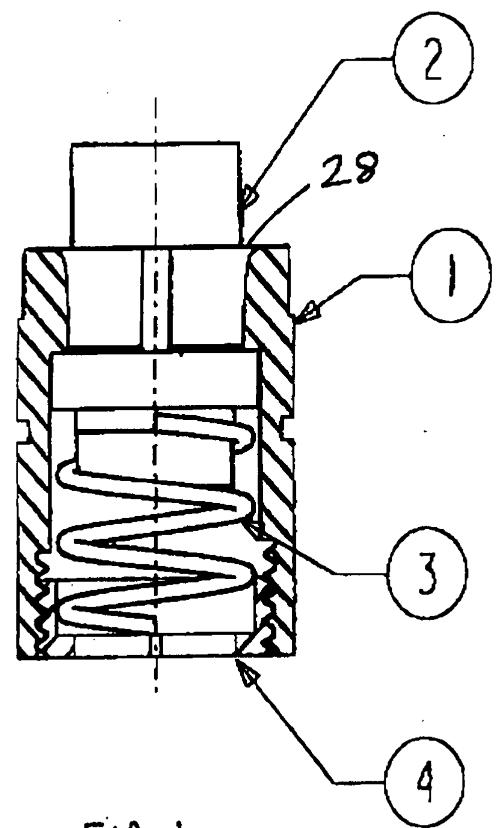

FIG.1

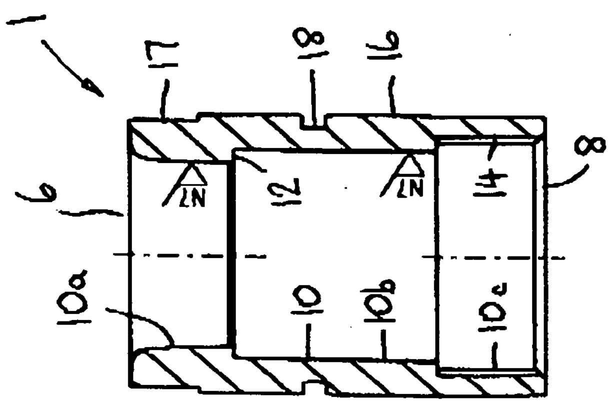

FIG. 2

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NOTE
MATERIAL:- STAINLESS STEEL
FINISH:- EXTERNAL SURFACES TO BE MACHINED TO GENERALTOLERANCE, UNLESS SPECIFIED.

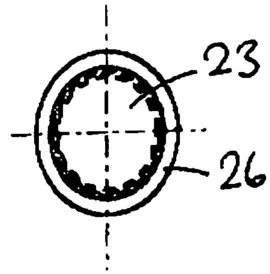
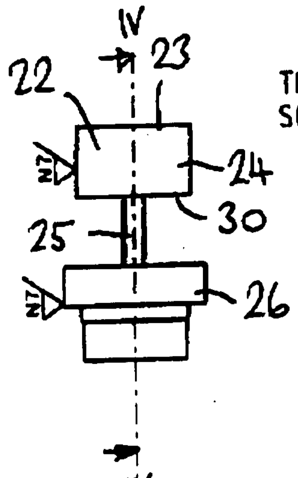


FIG.5



THESE EDGES TO BE SQUARE AND BURR FREE.

FIG.3

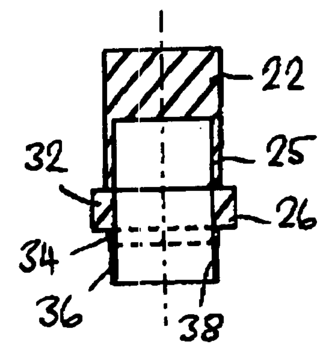
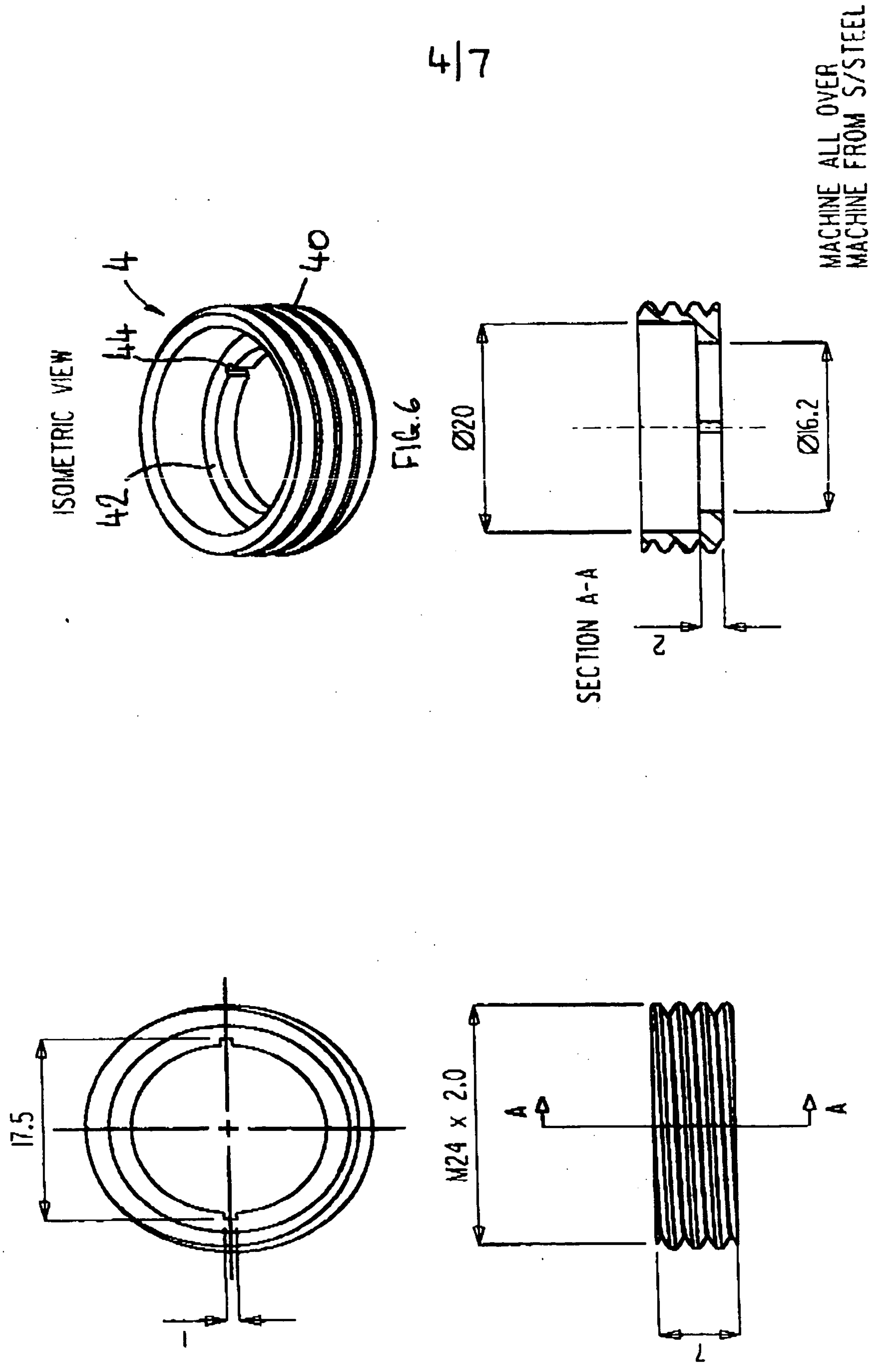
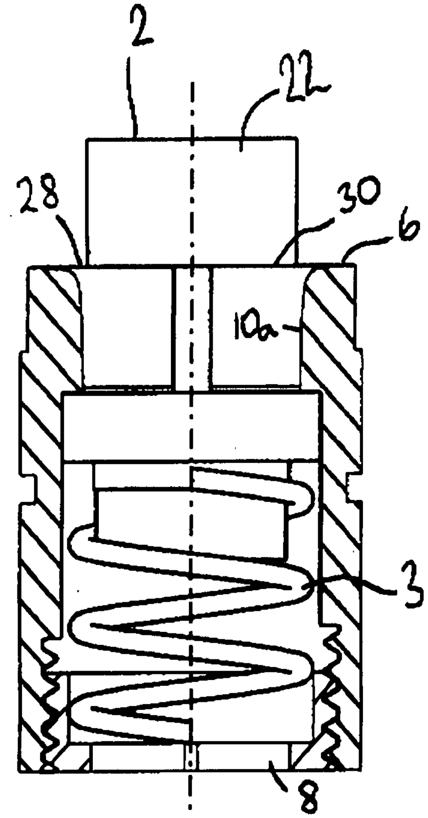


FIG.4

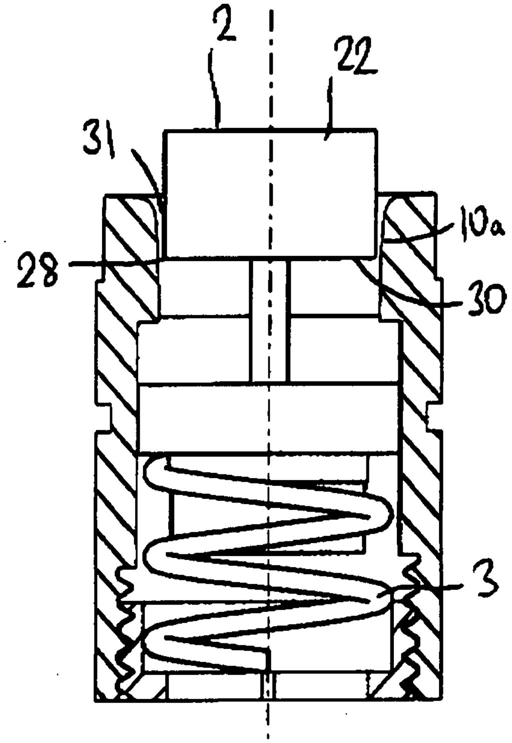

MACHINE ALL OVER
MACHINE FROM S/STEEL
ISOMETRIC VIEW
4
40
44
42
Ø20
Ø16.2
SECTION A-A
2
17.5
1
M24 x 2.0
A
A
7
FIG.6

5/7



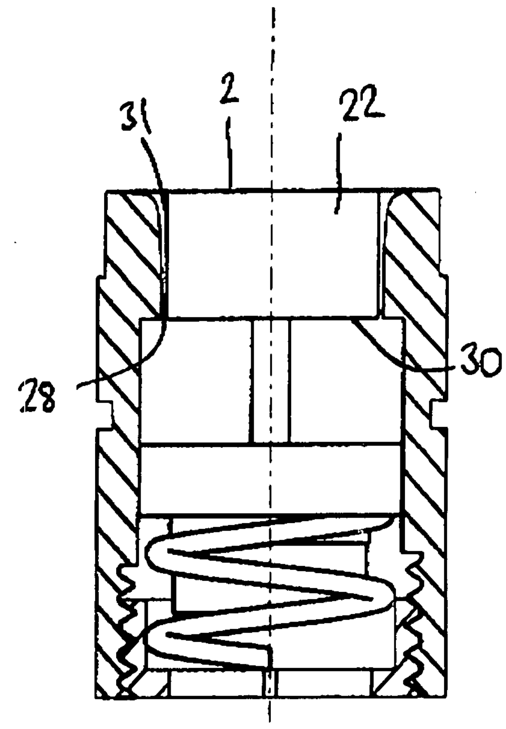
A) POSITION :- FULLY OPEN

Fig.7



B) POSITION :- INTERMEDIATE OPEN

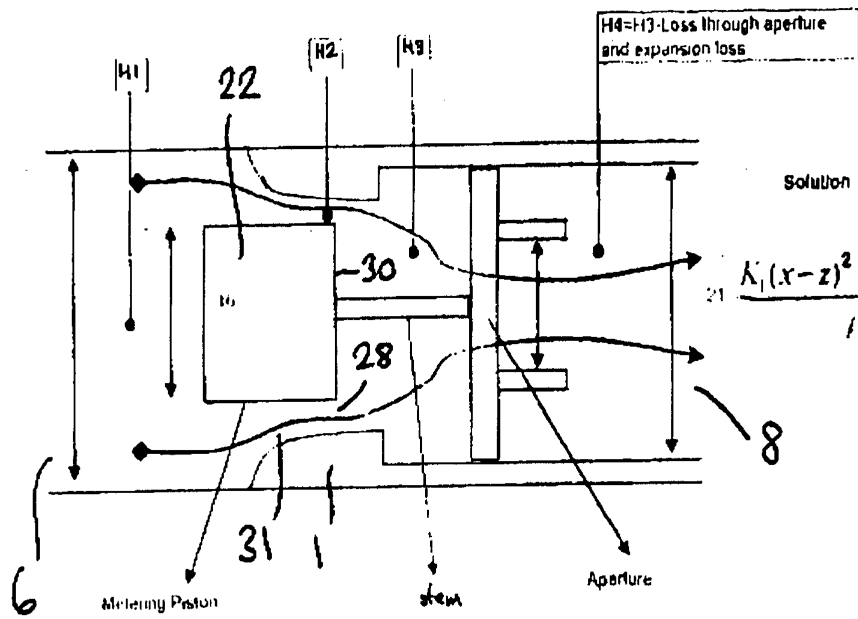
Fig.8



C) POSITION :- FULLY CLOSED

Fig.9

$$\frac{K_1(x-z)^2 + K_2(x-z)}{\rho g} = A_p(H_4 - H_3) - A_{base}(H_3 - H_4)$$



Solution

$$\frac{K_1(x-z)^2 + K_2(x-z)}{\rho g} = A_p(H_1 - H_4) - \frac{Q^2}{2gA_p^2}(A_p - A_b)(1 - \frac{A_p^2}{A_b^2})^2$$

Fig. 10

TYPICAL TRUMPET SIZES vs FLOW RATE

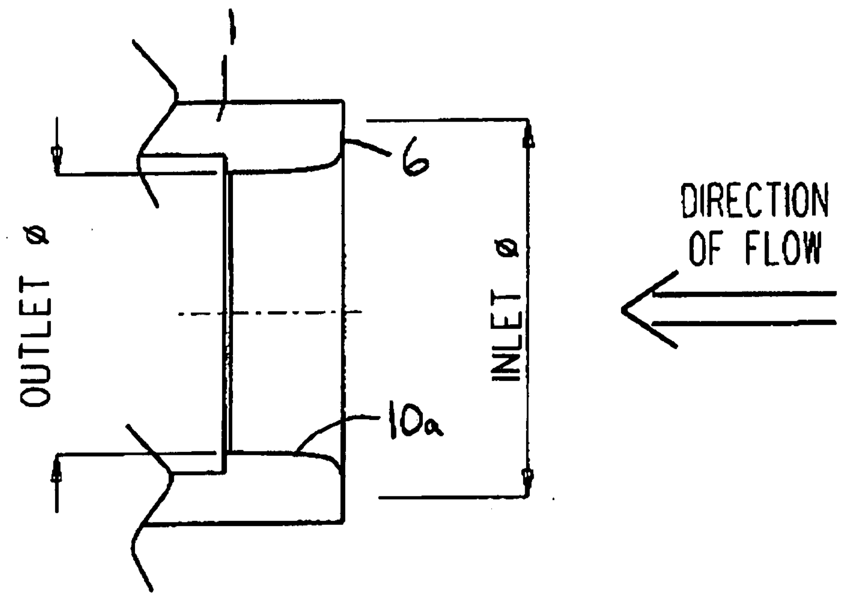


Fig. 11

SIZE	INLET DIA	OUTLET DIA	FLOW RATE l/s
3/4"	26.750	17.474	0.221
3/4"	26.626	19.059	1.199
1 1/4"	33.271	23.679	0.758
1 1/4"	31.422	24.357	1.263
2"	45.556	31.385	1.263
2"	45.575	33.343	3.157
3"	67.979	45.665	7.261